COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

Engineering Library

September, 1944

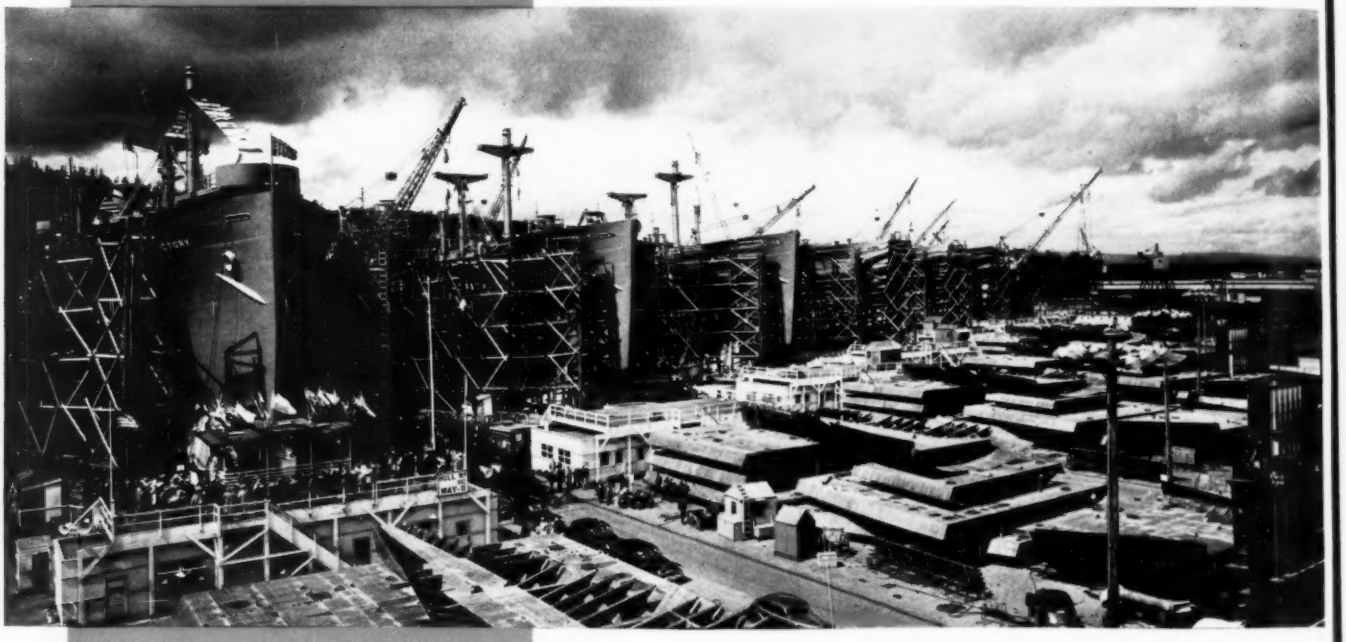
OCT 3 1944

Tandem low-pressure turbine of new 147,000-kw unit at Commonwealth Edison Company's Fisk Station; see page 38

147,000-Kw Unit at Fisk Station of Commonwealth Edison Company

Future Possibilities of Steam Power

. . . AND THEY'RE ALL POWERED WITH C-E BOILERS



Left to right

DOMINICAN VICTORY
SOUTH AFRICA VICTORY
YUGOSLAVIA VICTORY
PLYMOUTH VICTORY
NIANTIC VICTORY
ROCKLAND VICTORY
CLAREMONT VICTORY
CRANSTON VICTORY
RUTLAND VICTORY
ELMIRA VICTORY
MARSHFIELD VICTORY

STRETCHING farther than the lens of the camera can see, here, in various stages of completion, are eleven of the many Victory Ships being built by Oregon Shipbuilding Corporation. Since this photo was taken, every one of these ships has been launched. As they go to sea, increasing the enormous tonnage of our merchant fleet, all eleven of them, like a large percentage of their sister ships in The Victory Fleet, will be powered to their destinations by boilers designed and built by Combustion Engineering.

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DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

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VOLUME SIXTEEN NUMBER THREE

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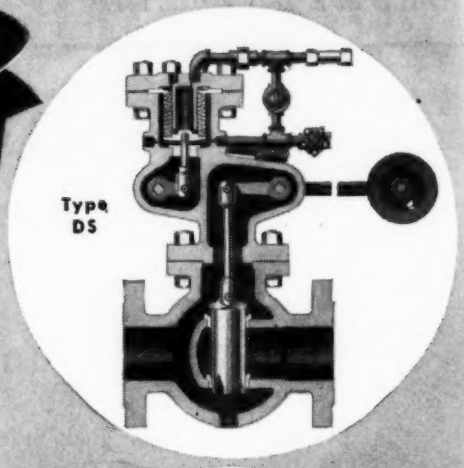

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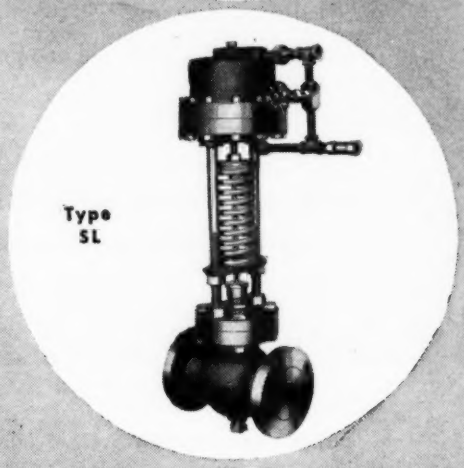

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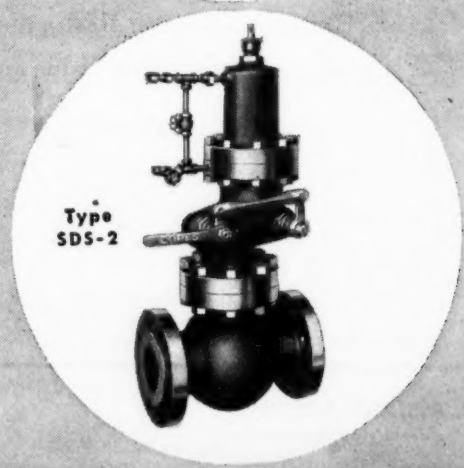

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EDITORIAL

Super-regeneration

Unlike some lines of production directly identified with prosecution of the war, in which advances in technique have been greatly accelerated, the field of power generation has been concerned mainly with the problem of supplying adequate and reliable power for war production. As such, the application of innovations has not been in order. This does not mean, however, that research and studies aimed at further improvement have remained dormant. A familiar example is the renewed interest in and the extensive development work that has been in progress with the gas turbine. At the same time, much active thinking is being directed toward further improvement in steam generation, such as may take advantage of war experience in the metallurgical and other fields and fit into post-war economy. Similar studies are being made by the British who are spurred by the prospect of more expensive coal than heretofore.

In this connection, the suggestion of J. F. Field, well-known British engineer, for improvement in the steam cycle by multiple regeneration, as abstracted elsewhere in this issue, is of interest. His idea of a steam "super-regenerator" may, at first glance, seem fantastic from a practical standpoint, but so have many other schemes when first proposed. It certainly represents a radical departure from present practice, but, on paper at least, it holds promise of substantial improvement over the heat rates of the most efficient installations now in operation. The proposal warrants constructive discussion.

Easement of Power Restrictions

Although the green light has not yet been given to new power construction, easement of the previous rigid restrictions is indicated by announcement that the Office of War Utilities is prepared to permit, under assigned rating, limited construction where present reserve capacity is inadequate, substantial fuel savings can be shown by replacement of obsolete equipment, or where extreme hardships exist. These considerations are understood to apply not only to power-generating equipment but also to transmission and distribution facilities.

Such action will, of course, be contingent upon availability of the necessary materials, noninterference with war orders, and upon the manpower situation in the particular locality. In fact, in certain areas of designated manpower shortage it is understood that construction will have to be carried out with existing labor forces.

Manufacturing facilities for producing power-generating equipment were greatly expanded shortly after we entered the war in order to meet the vast demands of war plants, as well as those of the Army, Navy and Maritime Commission. However, the construction stage of war plants has now been passed and there has been some tapering off in marine requirements, with the result that excess capacity is now available among many equipment manufacturers. Moreover, the materials situation has eased considerably in many respects.

While the demands for munitions and other implements of war are likely to be heavy for some time and may increase in certain categories, it is improbable that they will affect the excess manufacturing capacity mentioned or make necessary a reversion to previous restrictions concerning materials.

Source of Black Iron Oxide and Hydrogen in Boiler Water

Some years ago deposits of black iron oxide on non-ferrous turbine blading, supplied with steam from boiler water believed to be substantially free from oxygen, was attributed by some to dissociation of steam into oxygen and hydrogen at high temperature in the superheater. At the time it formed a subject of discussion among groups of engineers, the proponents of the idea basing their contention on the well-known fact that steam in the presence of iron will dissociate at high temperature. Subsequent research, however, at one of our universities showed that only extremely small amounts of oxygen and hydrogen are thus liberated at steam temperatures within the range of power-plant practice.

Meanwhile, through refinements in field investigations, more information is constantly being accumulated as to what goes on within a boiler.

In one case it was recently noted that, despite deaeration and chemical oxygen scavenging, considerable quantities of black oxide of iron were present in the boiler water and deposited within the tubes in the form of a very fine dust. While it was first thought that this might have been produced in the boiler, further investigation and tests showed a very large part of the iron to have been brought into the boiler with the feedwater, the source being the last turbine stages, condenser boxes, pipe lines, pumps and heaters. Actual measurements showed progressive concentrations of iron at these locations.

The black powdery oxide as found deposited on sections removed from the tubes is quite different from the hard scaly oxide produced by overheating.

Similarly, hydrogen in the steam was traced to the feedwater, probably through the breaking up of ferrous hydroxide into black iron oxide, water and free hydrogen, with the rising temperature. As in the case of the iron, and starting with zero hydrogen at the outlet of the condenser, measurements showed relatively large incremental increases in parts per million at the points mentioned.

Undoubtedly, what was found in this instance will go far to explain similar occurrences in other cases, even though the operating conditions may differ.

147,000-Kw Unit At FISK STATION of Commonwealth Edison Company

This single machine, with its two steam generators, has more than twice the capacity of the 14 units and 112 boilers originally conceived for the station. When compared with the original units on the basis of floor area occupied the concentration of power is tenfold. Capacity per unit installed has been increased 30 times and performance has been improved 500 per cent.

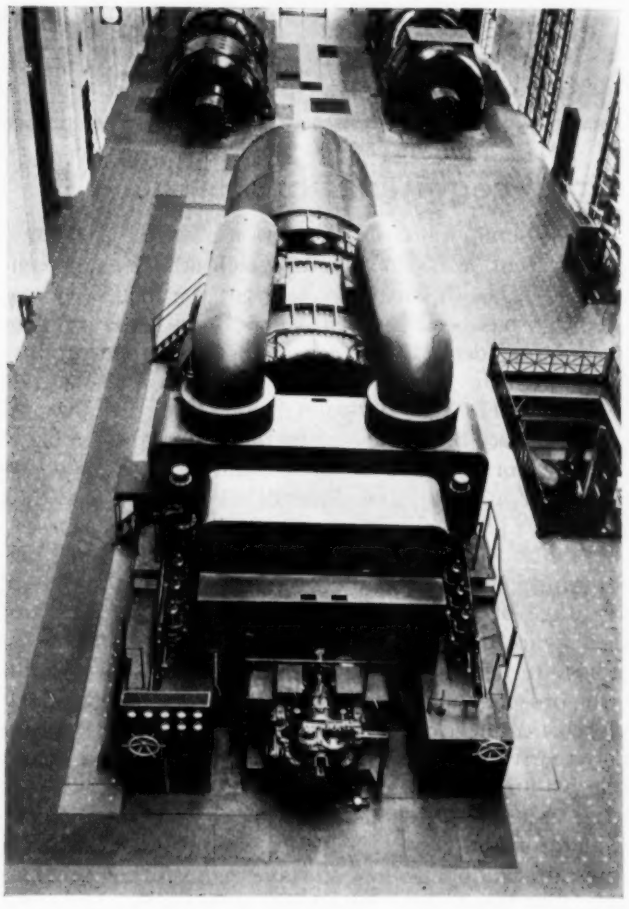

Fig. 1—New 147,000-kw turbine-generator at Fisk Station

OUTSTRIPPING in capacity any other turbine generator placed in service in this country since Pearl Harbor, the new 147,000-kw Unit No. 17 at Commonwealth Edison Company's Fisk Station is now running under full load and playing an important rôle in the job of meeting the huge power demands of Chicago's war production industries. It embraces the largest single-shaft turbine in the world operating at a steam pressure of 1250 psi and a temperature of 925 F.

This latest addition to the Chicago area power pool was installed at Fisk because of the station's proximity to the center of load and the economies made possible there through the replacement of three outmoded, inefficient machines. Adoption of 147,000 kw as the size for the unit was dictated by two factors—the space-saving advantage of its compact design and the estimated additional capacity requirements of the system as a whole. The operating steam pressure and temperature were chosen principally because of their desirability for an economical, modern, straight-through, simple condensing turbine.

The chronological sequence of the units installed at Fisk since the station was first placed in service in 1903 is shown in the accompanying table. To provide space for the new machine, vertical Units 8, 9 and 10 were removed. Room for the two steam generators and four feed pumps was afforded by removing the boilers originally installed for Units 6, 7 and 8. Thus the new 147,000-kw installation was placed in a station area originally provided, in 1903, for 15,000 kw of capacity. Actually it replaces obsolete units representing installed capacity of 36,000 kw.

Details of Turbine-Generator

Figs. 3 and 4, respectively, are cross-sections of the high- and low-pressure elements of the 1250-psi, 925-F, 1800-rpm Allis-Chalmers tandem-compound, condensing turbine unit. The steam path consists of a Curtis stage and thirty-nine reaction stages. Blades in the last row of the double-flow, low-pressure turbine have a tip speed of 1135 ft per sec. The throttle steam flow at full load is 1,262,000 lb per hr, while the condenser flow is 845,300 lb per hr, the difference of 416,700 lb being bled for feedwater heating. In all, five bleed points are provided. Three of these are located in the high-pressure turbine, one at the cross-over connection and one in the low-pressure turbine. A final feedwater temperature of 455 F is attained.

Because of the large amount of steam required, the turbine is provided with two steam chests, one on either side of the high-pressure cylinder. On each steam chest is mounted a 14-in. throttle valve and three 12-in. admission valves. An equalizing line connects the two steam chests to equalize flow at high loads. The turbine is designed for four valve points, two to the impulse stage, the third ahead of the reaction blading and the fourth bypassing six reaction stages.

The blade-carrying portion of the high-pressure spindle is machined from a solid single forging. A stub end is shrunk, keyed and bolted at the low-pressure end. The low-pressure spindle consists of a built-up welded assem-

FISK STATION INSTALLATIONS

Unit	Capacity (Kw)	Service Date	Make	Type	Throttle Steam Conditions: Press. (psig)	Throttle Steam Conditions: Temp. (F)	Boilers: Speed (rpm)	Boilers: No. per unit	Boilers: Capacity each lb per hr
Old No. 1	5,000	10- 2-03	G.E.	Vertical	200	513	750	8	25,000
Old No. 2	5,000	12-11-03	G.E.	Vertical	200	513	750	8	25,000
Old No. 3	5,000	4-21-04	G.E.	Vertical	200	513	750	8	25,000
Old No. 4	7,500	10-27-04	G.E.	Vertical	200	513	750	8	25,000
No. 5	12,000	9-17-06	G.E.	Vertical	200	513	750	8	25,000
No. 6	12,000	10-10-06	G.E.	Vertical	200	513	750	8	25,000
No. 7	12,000	11-13-06	G.E.	Vertical	200	513	750	8	25,000
No. 8	12,000	1-22-07	G.E.	Vertical	200	513	750	8	25,000
No. 9	12,000	7-19-07	G.E.	Vertical	200	513	750	8	25,000
No. 10	12,000	9-17-07	G.E.	Vertical	200	513	750	8	25,000
Replaced No. 4	12,000	5-21-09	G.E.	Vertical	200	513	750	.	....
Replaced No. 3	12,000	6-14-09	G.E.	Vertical	200	513	750	.	....
Replaced No. 2	12,000	9- 3-09	G.E.	Vertical	200	513	750	.	....
Replaced No. 1	12,000	10-28-09	G.E.	Vertical	200	513	750	.	....
No. 14	20,000	3-30-14	G.E.	Single cyl.	220	588	1200	4	90,000
No. 11	25,000	6-21-14	Parsons	Tand. comp'd.	220	588	750	4	80,000
No. 13	30,000	12-24-19	G.E.	Single cyl.	220	588	1800	4	90,000
No. 12	35,000	1-18-20	West.	Tand. comp'd.	220	588	1500	4	80,000
No. 15	30,000	10- 8-37	A.C.	Single cyl.	1200	900	1500	2	375,000
No. 17	147,000	12-11-43	A.C.	Tand. comp'd.	1250	925	1800	2	750,000

bly of forged-steel disks and two forged-steel shaft ends. This construction, unique in design, is used to insure a sound structure in the size of spindle involved.

The generator is a 147,000-kw, 60-cycle, 12,600-volt, three-phase, air-cooled unit. It is designed for 85 per cent power factor operation. Because of the machine's large capacity both in kilowatts and amperes, its armature has two electrically independent windings, each of which connects to a separate bus system. The terminals of the two windings are brought out on opposite ends of the armature to relieve congestion of cables under the machines as much as possible.

Because of the large size of the field, its core is built up of a forged-steel shaft provided with axial, serrated grooves into which teeth are fitted to form the sides of the coil slots and the center of the poles. This construction avoids the extremely large forging that would be required if one-piece construction were used as on smaller machines.

Steam Supply

Steam is furnished at 1325 psi, 935 F by two 750,000-lb per hr steam generating units. They are of the B. & W. high-head type, equipped with continuous-tube drainable superheaters, continuous-tube economizers, tubular air heaters, inter-tube type attemperators, water-cooled furnace walls and continuous slag-tap furnaces. Relatively low furnace heat release and wide tube spacings in the convection banks and superheaters were stipulated to prevent slag fouling. As a result, effective slag removal is attained with the mechanical blower installation which is composed entirely of retractable blowers in the boilers, superheater and economizer sections.

The fuel fired, in pulverized form, is Central Illinois coal, which has an ash content of approximately 14 per cent. Electric precipitators, one for each boiler unit, are located in the gas duct between the air heaters and induced-draft fans. They collect 90 per cent of the fly ash borne by the gases before discharging to the stack.

Boiler Control System

The automatic boiler control system is designed to provide three methods of control—full automatic, boiler manual and hand. Under full automatic control the system is actuated by changes in main steam-header pressure through a master pressure element controlling the regulators of both boilers. With boiler manual control the regulators of either boiler may be controlled by a single control element. Under hand control the regulators of each boiler may be controlled individually from the boiler board.

Control of the steam passing through and bypassing the attemperator of each boiler is effected only by a remote hand-control unit to regulate final steam temperature. Efforts to attain simplicity in the control system have resulted in a reliable system, although the installation is complicated by the fact that two induced-draft fans, two forced-draft fans and three sets of mills and feeders per boiler may be operated in various combinations. The control system is arranged to accommodate these operating requirements.

Four 600,000-lb per hr, 1600-psi boiler feed pumps are provided for the installation. Three are driven by

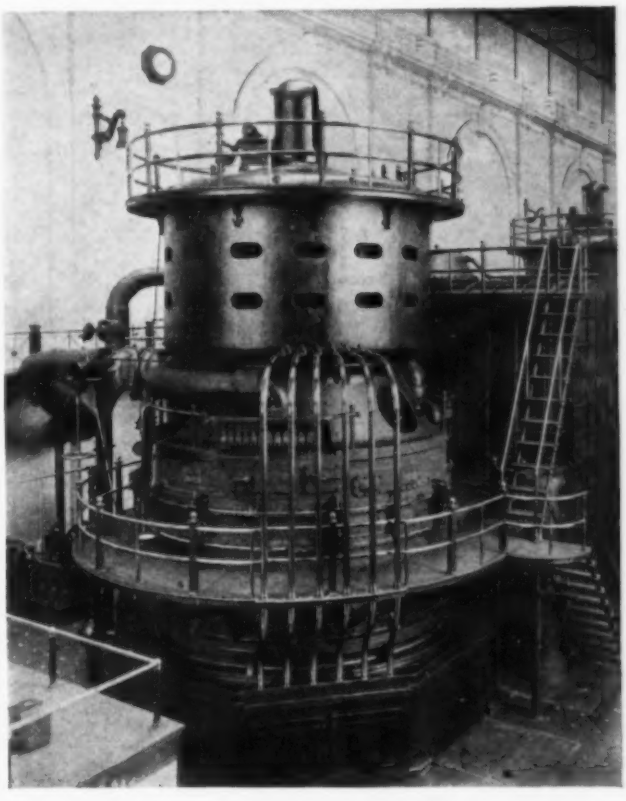

Fig. 2—One of the early vertical turbine-generators removed to make room for the new unit

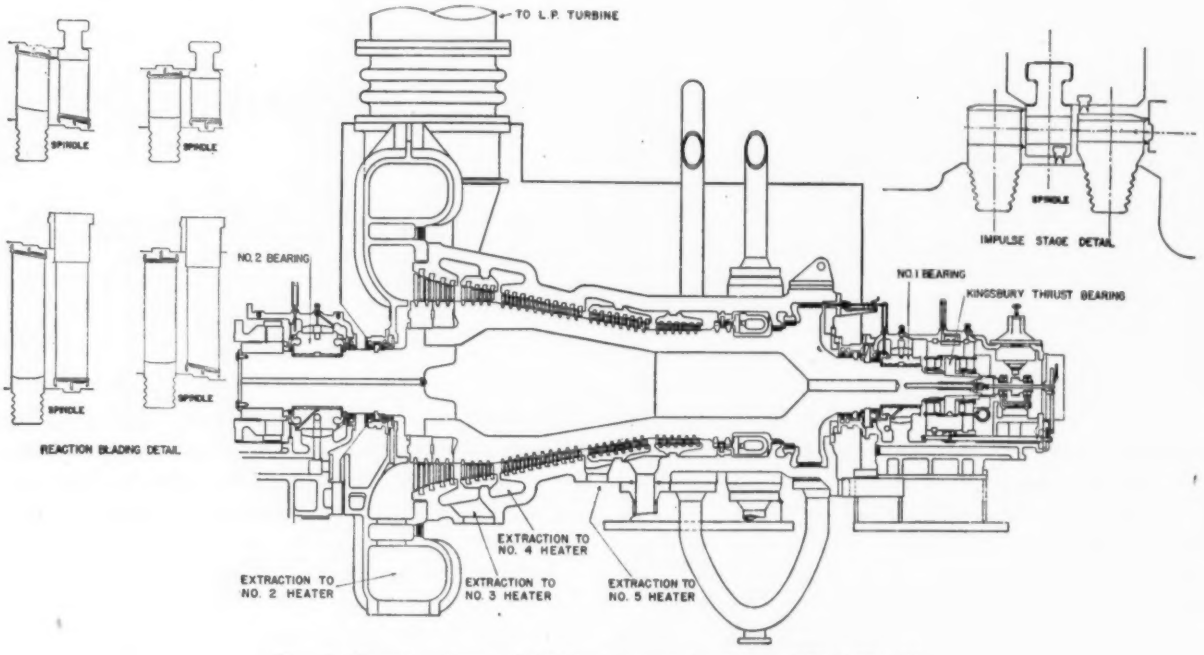

Fig. 3—Section through high-pressure turbine of Unit No. 17

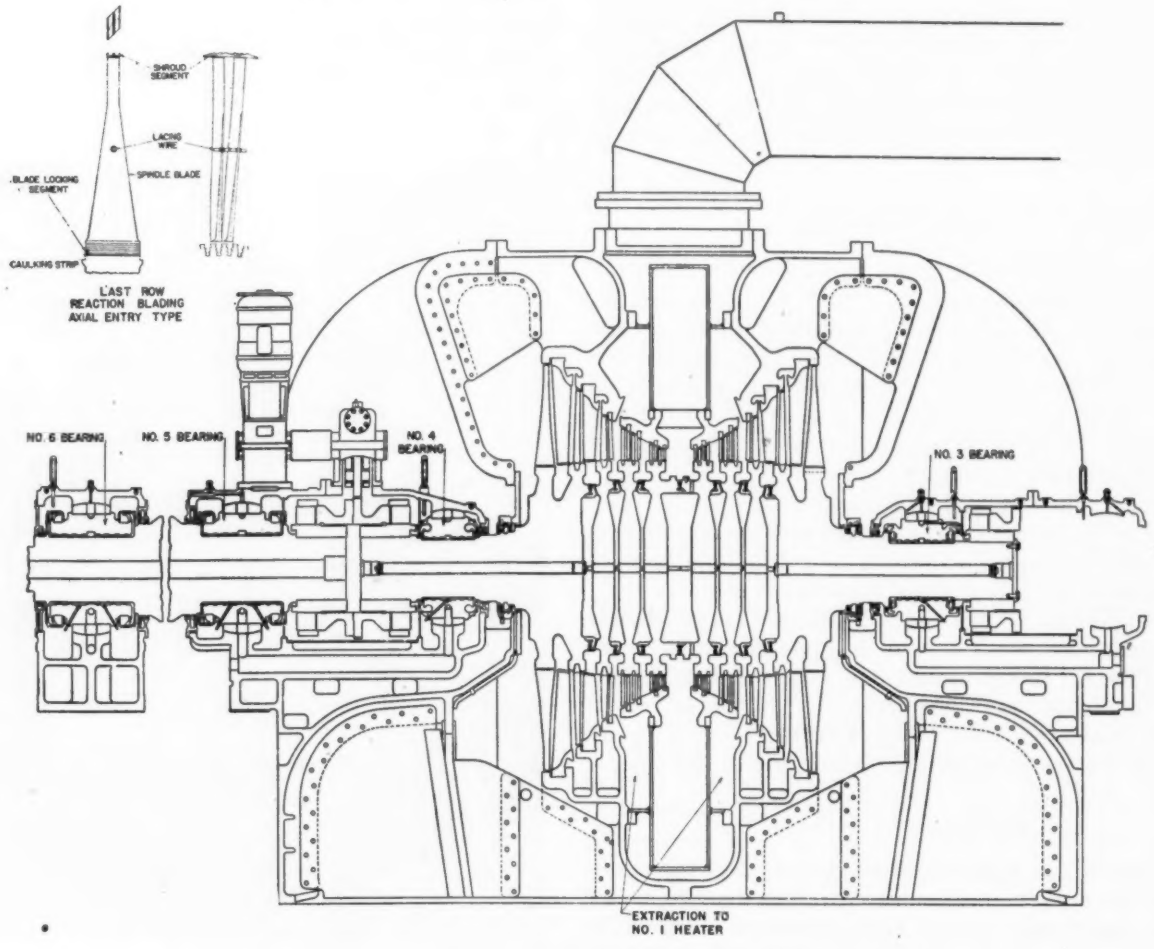

Fig. 4—Section through low-pressure turbine of Unit No. 17

constant-speed 1750-hp, 3570-rpm motors and one by a 1540-hp, 3340-rpm turbine. Feed pressure is controlled manually from the motor-driven pumps by throttling the pump discharge valves. Control with the turbine-driven pump is attained by regulating its speed. Normally, the turbine-driven pump is used as a spare.

With the installation of Unit No. 17, the station coal-handling system was modernized. The new plant has a capacity of 800 tons per hour and is considered ample to serve both present and proposed units at Fisk.

Turbine Auxiliaries

The turbine is provided with a 90,000-sq ft welded-steel-shell, radial-flow, single-water-pass condenser, equipped with arsenical admiralty tubes. Circulating water is supplied by two 54-in. motor-driven, vertical-shaft propeller pumps, each having a capacity of 82,500 gpm. These are located away from the unit in the station screen house. Three 5-stage, motor-driven, 1050-gpm, condensate pumps deliver condensate at approximately 350 psi pressure through the feedwater heating system to the suction of the boiler feed pumps.

Fig. 5 represents the flow diagram of the unit. Condensate is pumped from the deaerating storage hotwell at the bottom of the condenser through the feedwater heating system to the boilers. The feedwater heating system consists of five vertical feedwater heaters, a drain cooler, a CO_2 boiler, a vent condenser and a heat exchanger. Condensate from the condenser hotwell is discharged at 350 psi from the hotwell pumps through Nos. 1, 2, 3 and 4 heaters and drain cooler to the suction header of the boiler feed pumps. These pumps discharge at 1600 psi through the No. 5, or high-pressure heater, to the high-pressure feed header.

Heaters Nos. 1, 2 and 4 have drain pumps which discharge to the main feedwater line on the discharge side of the heater served. The drain from No. 3, which functions both as an extraction heater and evaporator condenser, passes through a drainer to the CO_2 boiler. A drain pump discharges the flow from the CO_2 boiler to the main feedwater line on the outlet side of No. 3 heater. The drain from the high-pressure heater passes through the coils of the drain cooler and then to the steam space of No. 4 heater.

Condensed steam from the evaporator coils passes through a drainer into the CO_2 boiler where it mixes with the drainage (evaporator distillate) of No. 3 heater. The condensed steam from the evaporator coils, being at a higher temperature than the drainage from the No. 3 heater, causes flashing in the CO_2 boiler, liberating dissolved gases from the makeup before it enters the feedwater stream. The vapor and gases driven off in the CO_2 boiler pass to the vent condenser where the vapor is condensed by the raw water going to the evaporator and the free gases are vented to the atmosphere. The condensed vapor from the CO_2 boiler-vent condenser passes to the heat exchanger, giving up heat to the raw water passing through the tubes, and is finally discharged to the reservoir. The amount of raw water evaporated constitutes the required makeup of the cycle.

The turbine heat rate at full load is 8750 Btu per kwhr gross. The auxiliary power at full load amounts to about 7000 kw so that with a boiler room efficiency of 87 per cent the overall net station heat rate will be about 11,000 Btu per net kwhr.

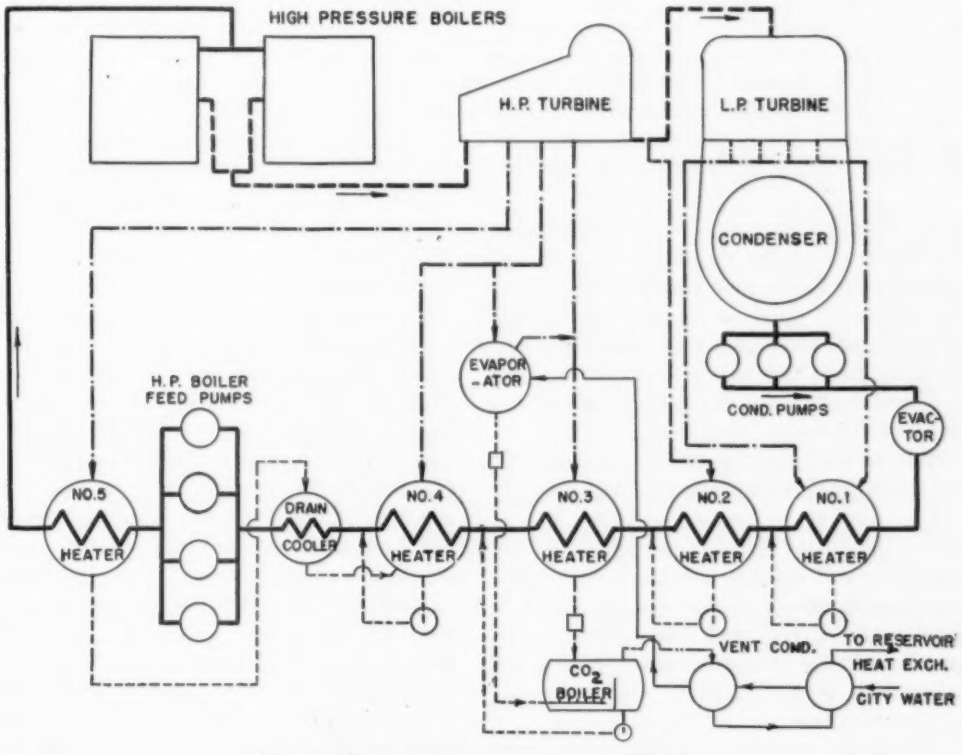

Fig. 5—Flow diagram of new installation

Use of Jets to Produce Turbulence in Spreader-Stoker Firing

By H. G. MEISSNER and M. O. FUNK

Combustion Engineering Company, Inc.

A résumé of the results of investigations made to determine the most favorable pressures for overfire air injection with this type of firing and comparison with the use of steam jets to avoid objectionable cinder carryover and smoke.

THE spreader stoker employs overfeed firing, the green coal being spread over the already burning fuel bed. The fines burn in suspension and the larger pieces are consumed on the grate.

This method of firing has many excellent features, among which are ability to burn a wide variety of coals in a simple and relatively inexpensive furnace—coals which would not be considered for use with other types of stokers. However, it also involves certain problems which should be understood if the good features are to be fully exploited.

As the stoker is generally installed in a vertical-wall furnace, there is little, if any, mechanical mixing or turbulence; hence, combustion conditions must be carefully controlled to prevent much of the cinder from the suspension burning being carried into the boiler passes. Also, as the green fuel is thrown over the incandescent fuel bed, the liberation of volatile hydrocarbons is rapid and it is necessary that these gases be quickly and completely mixed with the combustion air to avoid smoke passing out of the furnace. These points have been the subject of much investigation some of which will here be discussed.

The first step in this investigation was to determine the amount and character of the cinder carryover and the source and density of the smoke under a variety of fuel and load conditions. Tests were run at several plants with different types of boilers and, wherever possible, the results have been put in curve form for convenience. Figs. 1 and 2 show the layouts of two of the boilers from which test data were obtained.

It was found that smoke and fly ash are companion occurrences, and that which cures the one will usually cure the other.

It might be noted here that there is no substitute for a well-designed furnace and correct combustion control. The first place to stop smoke is at the source, which is in and immediately above the fuel bed. With good fuel-bed conditions and the proper fuel-air ratio, the battle is more than half won. Without these starting conditions, little can be done in the furnace that is of material help. Adequate furnace volume and flame travel are of prime importance.

In general, sufficient air can be supplied through the thin fuel bed, which is characteristic of this type of firing, and mixing or turbulence, rather than additional air, is desirable.

Tests were run with a number of different jet arrangements in some of which air was induced by means of steam jets, while in others small high-pressure fans were employed. The best overall results were obtained by the use of steam jets and no additional overfire air, practically all of the combustion air being supplied through the fuel bed.

In the words of one report, "observation of operation with steam jets which were entraining air indicated that an excessive amount of air was required over the fire and that in order to maintain reasonable CO_2 at the boiler outlet, the air pressure under the fire and the consequent flow of air through the fire had to be reduced to such an extent that the fuel beds clinkered badly and did not burn out properly. The entrained air was therefore eliminated and the steam jets alone were used to introduce turbulence in the furnace."

It is axiomatic in spreader-stoker firing that the fuel bed must be maintained in a thin, active and fully aerated condition. On the other hand, if the air flow through the furnace is excessive, then the furnace temperatures are reduced and incomplete combustion is likely to be experienced. Therefore, when air jets are used to induce turbulence, they should be designed for maximum velocity rather than for volume, so that the quantity of such air will be minimized. An estimate on the percentage of overfire air desirable is that it should not exceed about 5 to 6 per cent of the total air required.

It should be understood that the accompanying curves and tabulations show averages and trends rather than specific values, as the furnace design, water-cooled surface, flame travel and other factors largely affect the actual results.

In order to determine the percentage of fuel burned in suspension, the ash deposited in the various parts of the boiler and furnace was weighed and analyzed; whatever ash left the furnace, to be deposited in the boiler passes or stack, was considered to have come from coal which burned as it passed through the furnace and did not remain on the grate. On this basis the curves in Fig. 3 show the values obtained from low- and high-volatile Eastern and Midwestern coals. In one set of these curves, the effect of coal sizing is illustrated, the amount of coal which is burned in the furnace rather than on the grate, being in inverse proportion to the sizing, as would be expected. The analysis of the coal, particularly its ash content and volatile matter, appeared to have a marked effect, although there were, of course, other variables which could not be completely evaluated. In general, the suspension burning increases with the volatile content of the fuel.

The question as to how much of the fuel burns in sus-

pension is of interest as to its effect on the cinder carryover rather than for its effect on the boiler output or efficiency. Some coals, such as double-screened nut, may burn almost entirely on the grate, while other fuels, such as bug dust, may show as much as 40 to 50 per cent burned in suspension. Yet in either case full capacity and good efficiency may be obtained.

To answer another frequently raised question, regarding the amount of cinders which leave the furnace and enter the boiler passes as well as the quantity of cinders or fly ash leaving the boiler and entering the stack, the values shown in the curves in Figs. 4, 5 and 6 were determined. It will be noted that the effect of sizing is much more pronounced than that of coal analysis, although the cinder carryover is affected by the volatile in the fuel, it being generally greater when the volatile content is low.

From these curves it is apparent that there is an optimum combustion rate as regards the cinder carryover below which the combustion rate of the cinders is retarded because of the low furnace temperature and above which the effect of high gas velocities becomes the controlling factor. This point varies considerably with the coal being burned, although other factors such as furnace-heat release, flame travel and water-cooling surface undoubtedly affect the results materially.

The cinders leaving the furnace are relatively coarse and precipitate out of the gas stream whenever there is a change in direction or a drop in velocity. Full advantage of this characteristic is taken in the boiler passes by installation of hoppers and removal equipment, or by collection equipment outside the boiler in the form of hoppers, precipitators or cyclones.

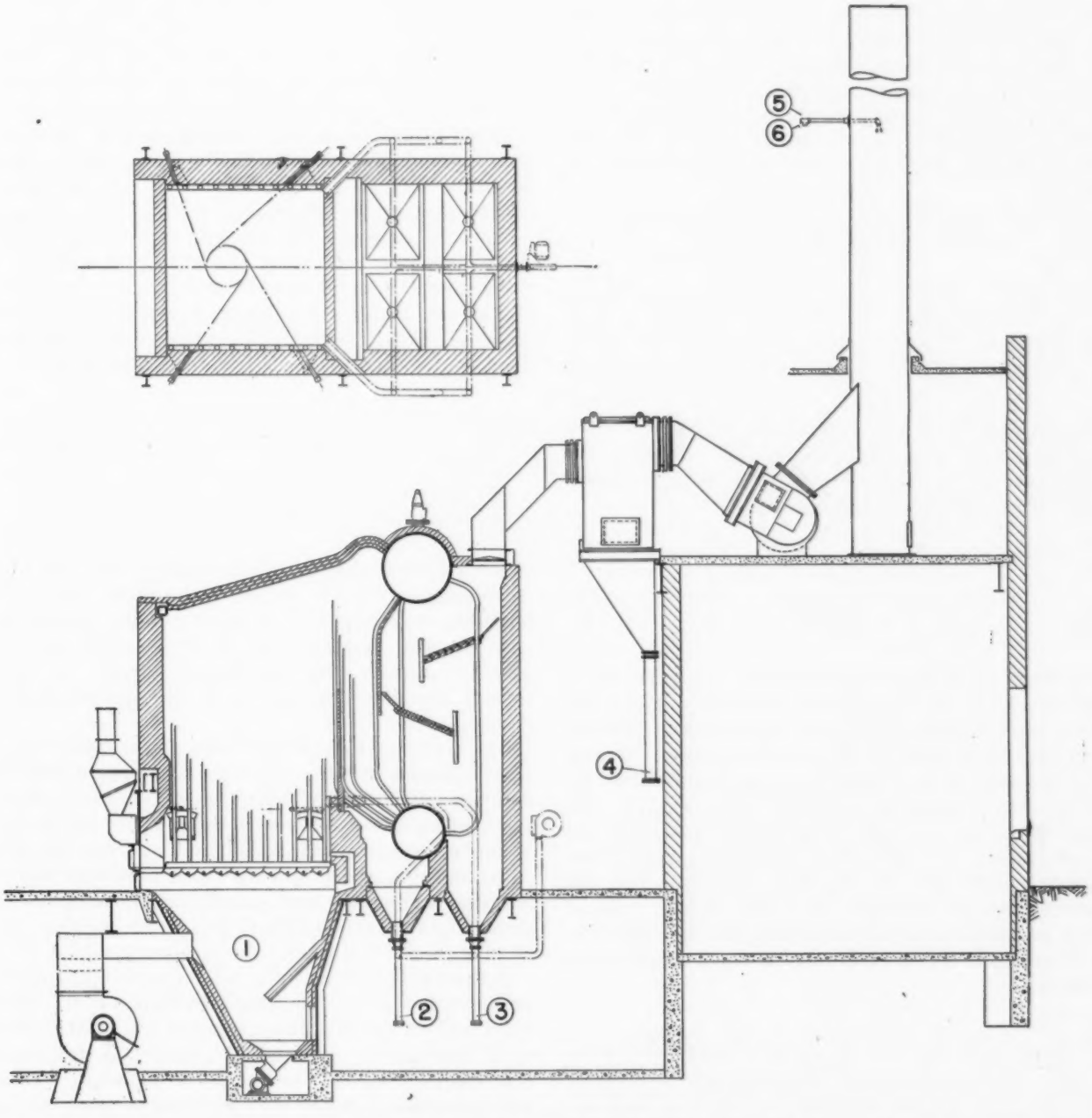

Fig. 1—Sketch showing location of cinder collection points in 2-drum boiler and recommended arrangement of steam jets

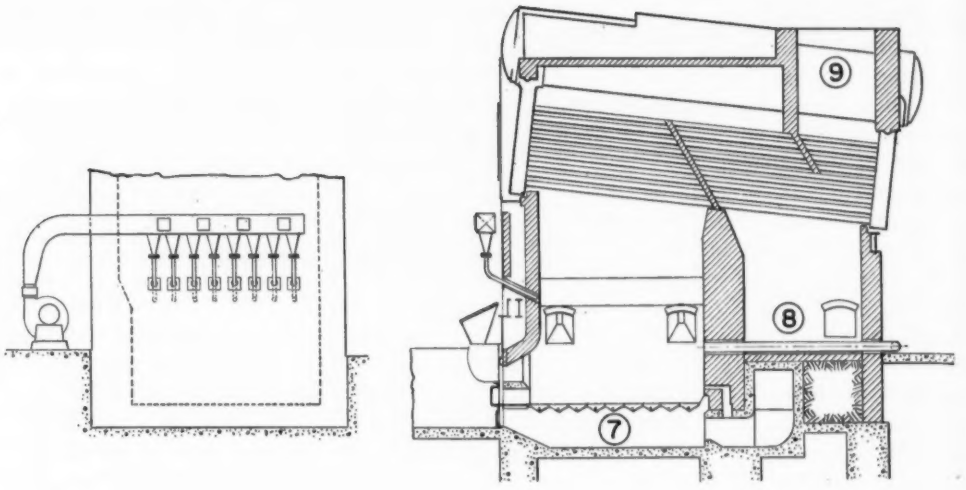

Fig. 2—Sketch showing location of cinder collection points in box-header boiler and typical arrangement of overfire air jets

The screen sizes of the cinders collected in the various sampling points are shown in Table 1. It will be noted that the majority of the coarser particles are collected in the boiler passes and precipitator. That which enters the stack contains relatively little in the nuisance range, which is sometimes considered as that held on a 100-mesh screen, although there is no general agreement on this point.

TABLE 1—SCREEN SIZE OF CINDERS

Samples Collected (Tyler Screen)	Per Cent of Cinders on Various Screens			
	In Boiler Passes	Leaving Boiler	In Precipitator	In Stack
On 14 mesh	0.58	0.00	0.00	0.00
On 48 mesh	63.72	10.72	11.96	9.48
On 100 mesh	17.83	13.30	15.62	10.86
On 200 mesh	10.45	25.19	19.29	31.10
Through 200 mesh	7.42	50.79	53.02	48.56

The combustible in the cinders collected at the various sampling points is shown in Table 2. These values are averages of a number of tests, there being no clearly marked trend as between different fuels.

TABLE 2—COMBUSTIBLE MATTER IN CINDERS

	Combustible in Cinders, Per Cent of Total Refuse			
	In Boiler Passes	Leaving Boiler	In Precipitator	In Stack
Range	40–85	30–75	30–50	35–75
Average	60.63	48.07	40.38	55.77

The tests on Indiana coal were run on a two-drum boiler rated at 25,000 lb per hr and with a water-wall furnace similar to that in Fig. 1. The cinders were collected from shallow pits in front of and behind the mud drum, as well as in a sampling pipe installed in the vertical run of breeching at the boiler outlet. The ashpit refuse was carefully weighed and analyzed, and the ash collected in the various sampling points was checked against that by the coal analysis. This ash balance[1] came out with an "unaccounted for" value of as little as one per cent when the work was carefully done and proved to be a good indication of the accuracy of the test results.

[1] The most difficult item to collect and sample by the above method is that part of the cinders which leaves the stack. By carefully weighing and analyzing the coal burned over a given period, the weight of ash fed to the furnace is determined. The refuse collected in the ashpit and the cinders picked up in the boiler passes, cinder traps and stack base, etc., are then weighed and analyzed. By difference the ash leaving the stack is then calculated, and by analyzing a sample taken from the base of the stack, a quite accurate value of the total cinders leaving the stack is determined. From the average CO_2 the total weight of gas can be calculated. It then becomes a simple matter to work out the dust loading for the test period.

Several sizes of coal, all from the same field, were burned at different rates. As noted on the curves, the sizes ranged from $^1/_4$ in. × 0 to $1^1/_2$ in. × 0. With a little readjustment of the stoker, the various sizes were distributed and burned satisfactorily, the principal adjustment required, being a change in the distributor-shaft speed from 600 rpm with the $^1/_4$-in. fuel to 400 rpm when the $1^1/_2$-in. × 0 size was fired. Average analyses of the coal tested are shown in Table 3.

Another series of tests were run on a two-drum boiler with a water-cooled furnace similar to that shown in Fig. 1, burning first an Ohio coal and then a Pocahontas coal, the average analyses of which are shown in Table 3. These fuels were fired at several different rates and with various combinations of overfire air and steam nozzles in use.

TABLE 3—COAL ANALYSES, AVERAGE OF TEST SAMPLES

	Indiana 5 Vein	Ohio	Pocahontas
Moisture	15.35	4.62	4.06
Volatile	32.70	39.85	21.99
Fixed carbon	42.30	46.73	69.83
Ash	9.65	8.80	4.12
Btu as fired	10,500	12,780	14,450

Several preliminary runs were made to test out the air entrainment with the nozzles to be used. The results obtained were found to be substantially in agreement with the data published by R. B. Engdahl (see COMBUSTION, March 1944) as regards volume of air entrained and the effectiveness of the various steam jet designs.

The cinders which dropped out in the boiler hoppers were collected in 8-in. pipe sleeves having double slide dampers which were attached to the hoppers in place of the existing cinder recovery nozzles, as shown in Fig. 1. A similar arrangement was used under the precipitator.

To collect samples from the stack, a cyclone was used in order to procure a large sample for screen analysis and an alundum thimble was employed for the weight sample, all in accordance with standard test procedure.

A third series of tests was run on a straight-tube box-header boiler, Fig. 2, burning Pocahontas coal. In this furnace there was installed a series of jets pointing downward through the front wall. Overfire air for the jets was supplied at 14.2 in. wg by a high-speed blower. The boiler was operated both with and without these jets and measurements of the cinder carryover and smoke were recorded.

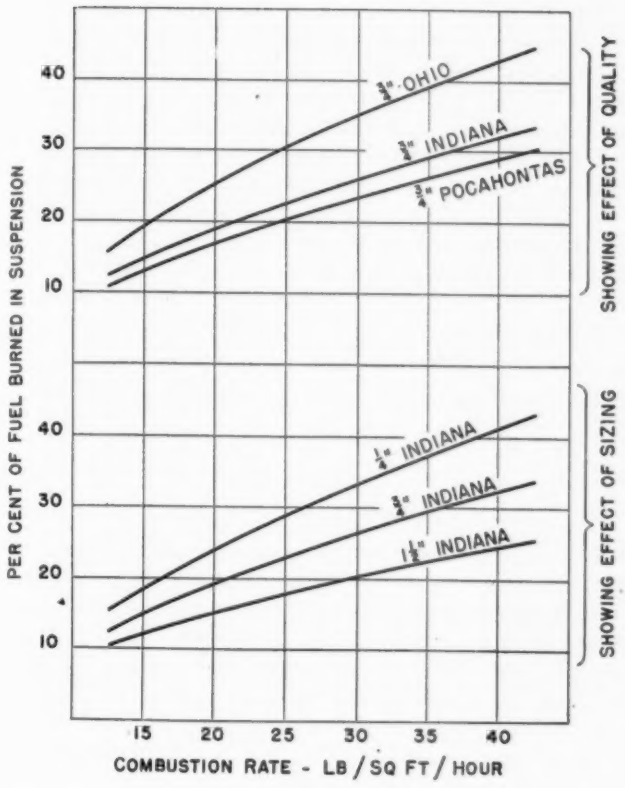

Fig. 3—Chart showing proportion of fuel burned in suspension

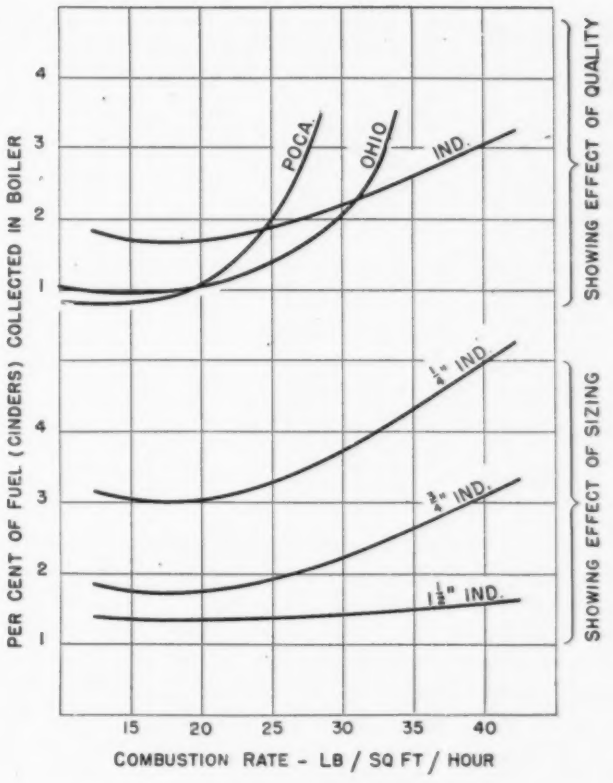

Fig. 5—Cinder collection chart showing effects of quality and sizing

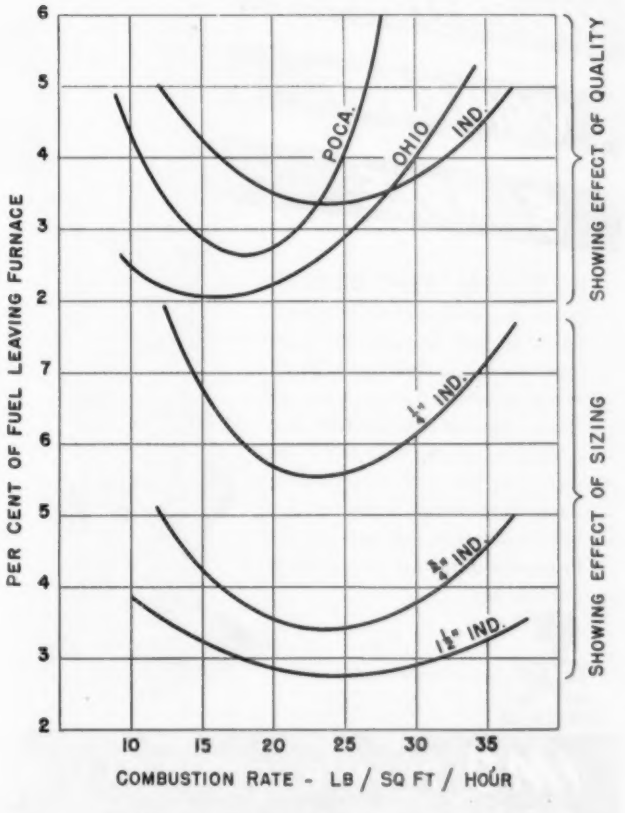

Fig. 4—Cinder emission chart

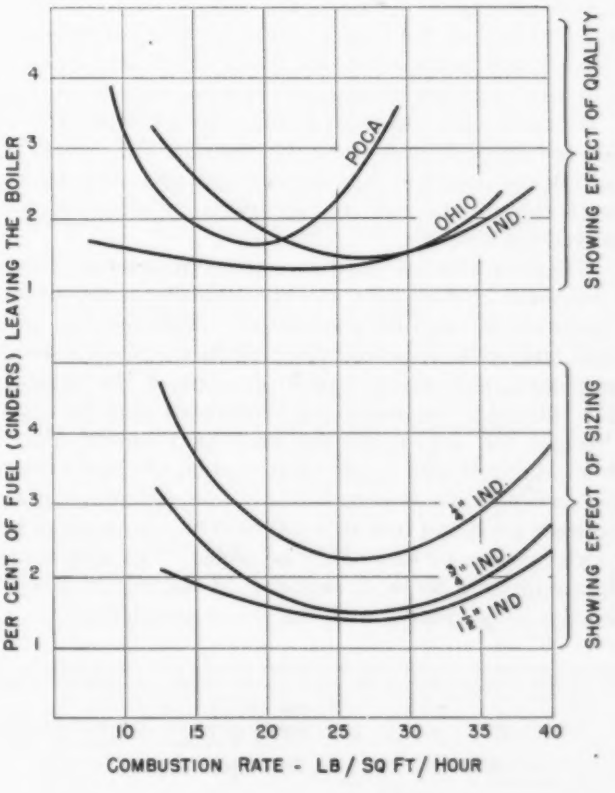

Fig. 6—Cinders leaving boiler in per cent of coal fired

TABLE 4—RELATIVE EFFECTIVENESS OF VARIOUS STEAM AND AIR NOZZLE ARRANGEMENTS ON BASIS OF CINDER AND SMOKE EMISSION, USING OHIO COAL. STEAM AT 115 PSI SATURATED. NOTE THAT ARRANGEMENT NO. 6 GAVE THE BEST PERFORMANCE AS REGARDS BOTH SMOKE AND CINDER EMISSION

	Relative Steam Consumption	Relative Emission	
		Smoke	Cinders
Arrangement No. 1—Two 1/8-in. steam nozzles at rear corners, each 34 in. above grate, inclined toward grate	1.9	2.16	1.59
Arrangement No. 2—Four 1/16-in. tangential steam nozzles at four corners of furnace each 34 in. above grate inclined toward grate	1.0	1.49	1.94
Arrangement No. 3—Same as arrangement 2 except that nozzles were 1/8 in. diam.	3.8	1.71	1.15
Arrangement No. 4—Three 1/8-in. steam nozzles equally spaced across front of furnace, 6 ft 6 in. above grate, inclined toward grate	2.9	4.51	1.43
Arrangement No. 5—No nozzles in furnace	0.0	5.60	1.82
Arrangement No. 6—Four 1/8-in. tangential steam nozzles at four corners of furnace each 34 in. above grate—horizontal	3.8	1.00	1.00
Arrangement No. 7—Eight 3/32-in. steam nozzles, 4 in each side wall, 5 ft 0 in. above grate, inclined toward grate	4.6	1.87	1.76

Smoke density in the stack was measured by mounting a spotlight over a hole in one side of the stack and a light meter on the opposite side. The light was focused upon the light meter and observation indicated that the amount of light as shown by the meter varied inversely as the amount of smoke in the stack. This proved to be a simple and reliable method much more accurate than the Ringelman chart, which depends so largely on the light conditions and the human element.

Dust loading in the stack gases was determined in some of these tests and it was found that this could be held well below the 0.75 grain per cu ft as set by the St. Louis and other smoke ordinances when the furnace conditions were correctly maintained.

Conclusions

The use of steam or air jets to promote furnace turbulence is frequently desirable, the effect of such turbulence being to reduce the cinder carryover and smoke and to increase the boiler efficiency by as much as 1 to 3 per cent when furnace conditions are otherwise satisfactory.

In many cases ample air is introduced through the fuel bed to completely consume the combustible gases, but turbulence is required to mix the gas and air. In such cases steam jets with no additional overfire air have proved satisfactory.

When air jets are used to induce turbulence, the arrangement of these jets, and the quantity of the air supplied must be carefully considered. High velocity rather than large volume is desirable and the overfire air should, in general, not exceed 5 or 6 per cent of the total air. The allowable percentage of overfire air may be readily checked for any particular fuel by reducing the air flow through the grate and noting at what point fuel-bed conditions deteriorate. The difference in CO_2 between good and bad fires will then be a measure of how much overfire air may safely be added. Air at pressures approximating 30 in. is necessary if the required turbulence is to be created with the above air quantity.

TABLE 5—RELATIVE ANNUAL OPERATING COSTS OF PRODUCING A GIVEN TURBULENCE BY MEANS OF VARIOUS TYPES OF JETS

	Per Cent of Total Combustion Air or Steam Required	Relative Annual Operating Cost
Fan operating at 5 in. S.P. Wg	12.0	1.0
Fan operating at 10 in. S.P. Wg	9.1	1.5
Fan operating at 30 in. S.P. Wg	5.7	2.9
Steam jet, no entrained air	1.1	3.6
Steam air ejector	0.5	1.8

The relative effectiveness of the various jet arrangements tested is shown in Table 4. It will be noted that the four tangential jets per arrangement No. 6, as illustrated in Fig. 1, proved to be the most effective as regards both smoke and cinder reduction.

Table 5 points out the relative values of the various methods of creating turbulence, showing that air jets of 5 in. to 10 in. S.P. are the most economical in operation, but that the air volume required is such that it may have detrimental effect on the fuel bed or overall efficiency.

TABLE 6—STEAM FLOW THROUGH VARIOUS SIZED NOZZLES, USING SATURATED STEAM. FLOW WITH SUPERHEATED STEAM IS SOMEWHAT LOWER

Steam Pressure, Lb/Sq In.	Steam Flow, Lb/Hr, Nozzle Diameter				
	1/16 In.	3/32 In.	1/8 In.	5/32 In.	3/16 In.
100	18	40	71	112	162
150	26	58	102	160	231
200	33	75	134	210	300
250	41	93	166	259	...
300	49	111	197	308	...

Steam air ejectors must be used with discretion for the same reason. Air at pressures of about 30 in. requires a minimum quantity, but is less economical to operate. Steam jets with no entrained air are shown to be the most expensive to operate, but are quite effective; and when used only as required have proved satisfactory.

In Table 6 is shown the steam consumption of various sized jets from which may be calculated the hourly rate for any given jet arrangement.

The correct arrangement and use of steam or air jets to create turbulence will, in general, increase the overall efficiency or capacity of the boiler sufficiently to more than balance the cost of operation.

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Future Possibilities of Steam Power*

By J. F. FIELD

City Electrical Engineer of Edinburgh

The first section of this paper (not here reproduced) reviews the probable limits of the present steam and mercury-steam cycles as represented by the most efficient installations in England and the United States, namely, Battersea A and B, Port Washington, Twin Branch and Kearny, and concludes that with existing technique the limit appears to have about been reached. The author proposes what he terms a steam "super-regenerator" employing a nearly reversible cycle with multiple regeneration up to the maximum operating temperature. The arrangement would comprise a specially built turbine, multiple regenerative heating, a directly fired high-pressure superheater, compressor and air preheater. The estimated thermal efficiencies range from 37 to 43.5 per cent, according to the maximum operating steam temperature.

THE requirement for reversible regeneration is that all expansion or compression should be as nearly as possible adiabatic or isothermal, and that bled steam should have as nearly as possible the same temperature as the feedwater to which it rejects its sensible or latent heat. This problem is comparatively easy in the saturated range of adiabatic expansion and is theoretically more efficient the greater the number of tapping points. To preserve reversibility in the superheat phase, however, the bled steam must be compressed isothermally. In practice, compressors can approach isothermal compression by division into a sufficiently large number of adiabatic stages with suitable inter-stage heat exchangers, as in high-pressure air-compressor technique; but first let us examine the thermodynamic problem.

Fig. 1 indicates a perfect regenerative steam cycle with an upper temperature of 1000 F and a lower temperature of some 79 F (a vacuum of 29 in. of mercury). It has the same efficiency as Carnot's compression cycle between the same limits of temperature, i.e., about 63 per cent. With an upper temperature of 850 F the theoretical figure would be about 59 per cent; and, with an upper temperature limit of 1200 F it would be about 68 per cent. Assuming an operating pressure of 3200 lb per sq in. (that is, the critical pressure) to facilitate the process of regeneration—although, in theory, any convenient pressure might be chosen—perfect regeneration requires that all heat from the bled steam be added to the feedwater at the same temperature as the feedwater. Above the critical temperature of some 700 F, the feedwater would become feed steam.

Assuming 100 deg F steps of regeneration, as in Fig. 2, to heat the feed steam reversibly from B to A it would be necessary so to compress bled steam from condition A_1 that it may fall along the condition line A_1–B or, if bled at point B_1, along the condition line B_1–A. The quantity of steam to be bled as a proportion of the whole is inversely proportional to the areas of heat subtended by AB and A_1-B or A-B_1.

Approach to reversible regeneration could be made by a sufficient number of adiabatic steps with suitable inter-stage feed heat exchangers, as shown in Fig. 3. Steam bled at B_1 may be compressed in a series of adiabatic stages with inter-stage heat rejection at gradually dropping temperatures to point C, in contraflow with the main feed steam, rising in temperature from point C to B. Alternatively, steam may be bled at point D with a similar adiabatic compression and inter-stage heat rejection, gradually rising in temperature to point C and rejecting heat to the main feed steam flow between D and C. In the saturated phase, one could compress or expand the bled steam, in the latter case doing useful work by stages of adiabatic expansion and stage-by-stage condensation on the feed heater surfaces.

Fig. 4 shows the condition line of the bled steam in Fig. 3, reduced to the appropriate scale for the total regeneration heat required by the cycle. The feedwater passes directly into steam at about 700 F, or at a lower temperature if a lower operating pressure than the critical is chosen. The boiler, as such, disappears.

The functional arrangement of such a steam plant is shown in Fig. 5. It is purely diagrammatic, and the actual multiple superheater and air preheater would probably be fairly bulky relative to the turbine. Also, the steam conditions given for the bleeder points correspond roughly with Figs. 2, 3, and 4, and no attempt has been made to allow for stage reheat. Any kind of compressor suitable to the pressures involved could be adapted, the main requirement being that a substantial number of adiabatic stages are introduced to achieve a reasonable degree of reversibility. It can be gathered from Fig. 1 that the throttle flow of steam is considerably larger than that rejected to the condenser, so that for a given size of steam turbine, the low-pressure blading and condenser would be smaller than in present practice, and the front end blading would be a good deal deeper. This would ease the designers' problem, particularly with machines of

* Excerpt from a paper appearing in the June 9 and 16, 1944, issues of *Engineering* (London) and here reproduced in part at the request of the author and through the courtesy of the editor of that publication.

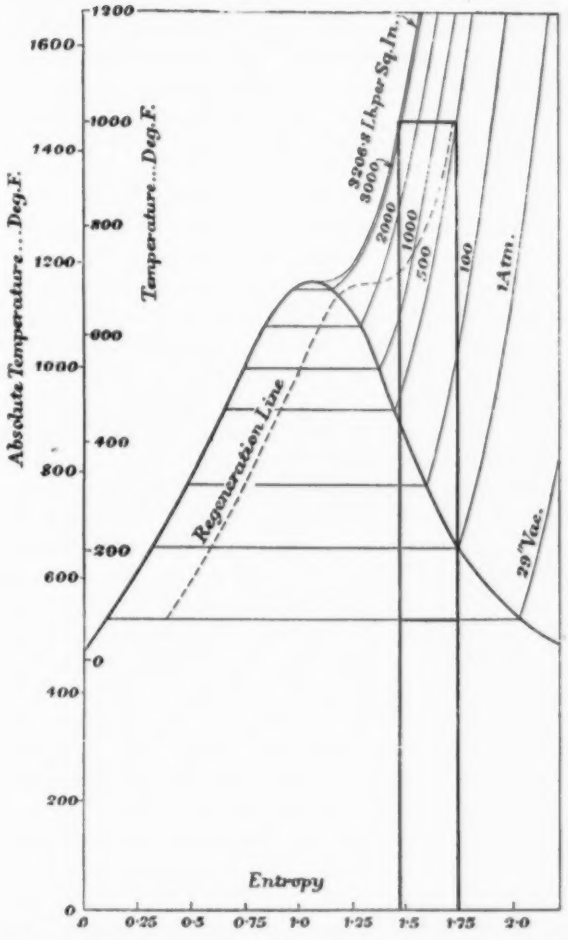

Fig. 1—Diagram of perfect regenerative steam cycle with 1000 F top temperature

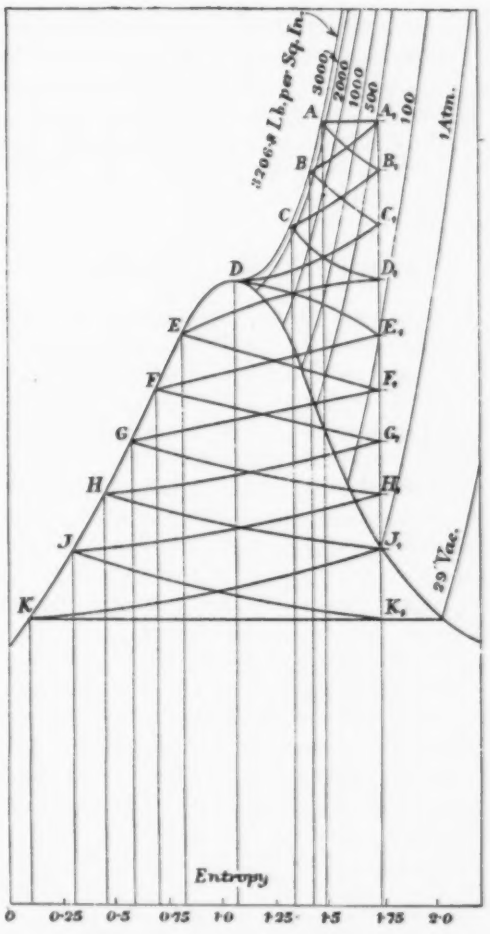

Fig. 2—Representing regeneration in 100-deg steps; steam conditions as in Fig. 1

moderate capacity, where, with existing heat cycles, it is sometimes difficult with high pressures to get deep enough blading to preserve a reasonable degree of high-pressure stage efficiency.

The high-pressure multiple superheater should give much less trouble than any kind of boiler, mercury or otherwise, due to the homogeneous condition of the fluid, although special control gear and emergency "soft" steam cooling might be necessary to avoid burnouts. A good deal would depend on the kind of throttle and fuel-control gear developed for the job. The thermal storage effect in such a plant would be virtually nil and thus the speed-load governor could be most effectively connected directly to the fuel and air supplies. Starting could be effected by introducing steam after the last reheat stage from a small auxiliary boiler of normal design, so that a suitable quantity of steam could be recirculated through the regenerator and reheaters, the furnace of the latter then being started up and the auxiliary boiler shut down. There are ample data available to design a highly efficient reheater turbine of the type envisaged, and the same may be said about the multiple superheater or reheater with regenerative air heater, so that the success of the scheme would depend on the thermodynamic reversibility of the dynamic feed heater-boiler. It is nothing more than a special kind of heat pump, and its efficiency criterion could be determined separately if necessary. It should become an important new organ in the technique of steam power, and could be termed a "super-regenerator."

The only other way of achieving the same theoretical efficiency with steam would be to expand the whole of it to the chosen back pressure, condense a small proportion, and recompress the mixture back to the starting point of the cycle; that is, a straight Carnot cycle arrangement except for multi-stage reheating, using the gas-turbine technique in which both turbine and compressor are several times the normal size of turbine for the output considered. Such an arrangement would require exceedingly high efficiencies in both turbine and compressor to avoid crippling losses.

Application to Existing Plants

The principle could also be adapted to existing medium-pressure plants of up-to-date design with an efficient arrangement of feed heating. The idea of superimposing a high-pressure back-pressure turbine on existing plants, with new high-pressure and high-temperature boilers to suit, can be modified to include the "super-regenerator" principle by designing the back-pressure superimposed element for multi-stage reheating, and arranging that a sufficient quantity of steam then passes out to feed the existing low-pressure set, which could remain on its existing site. Sufficient steam for pressure feed heating only would then be expanded through the remaining stages of the "super-regenerator" set, with stage-by-stage pass-out to the compressor-type feed heater, which could accept feedwater already heated by bleeding on the existing set. The superimposed plant, with dynamic

feed heater, would preferably be situated close to the multiple resuperheater, which takes the place of the ordinary high-pressure boiler. With a medium-temperature plant of good design, there should be a much greater increase, both in capacity and in thermal efficiency, than is possible with an ordinary back-pressure plant at any particular initial temperature.

Comparison with Gas Turbine

In appraising the above suggestions, it will be remembered that the whole family of heat engines, whether of the internal-combustion or external-combustion variety, may be considered to lie between the two extremes of the simple steam engine using a highly condensible working fluid, so that the negative work of the cycle is very small compared with the positive work, and the gas turbine, where the positive work may be about five times, and the negative work about four times, the net output. In the former, the efficiency of the feed pump or negative-work element scarcely matters, the efficiency of the heat engine as a whole depending almost directly on the efficiency of the positive-work element or turbine. In the gas turbine, the efficiencies of both negative-work and positive-work elements each has an almost *pro-rata* effect on overall thermal efficiency, and both this and the net output of available work deteriorate very rapidly below a certain point of relatively high efficiency for both elements.

The development of the steam turbine described in this article, viewed as a complete heat engine, lies between these two extremes, but is very much nearer to the former, for its negative work is still a mere fraction of the net output, since most of the negative phase of the cycle is still with the working fluid in the liquid state. Therein lies its overwhelming practical advantage over any kind of noncondensible gas turbine. High-compressor efficiency is important, for it is necessary to recover as much as possible of this recirculating energy in the positive-work phase, but the machine is a comparatively straightforward modification of the ordinary feed-heating steam turbine; and its heat-cycle efficiency from any particular upper limit of temperature is inherently very much higher than that of the gas turbine working from the same upper temperature, partly because the properties of the working fluid allow efficient expansion down to a much lower sink temperature and partly because the process of regeneration can be made much more nearly perfect. In short, the heat cycle has a much higher ideal thermal efficiency than that used in the gas turbine, and, because of the much smaller negative work, it should be much easier to make the heat engine as a whole more efficient in converting the available heat of the cycle to mechanical work. Every improvement in blading material for gas turbines could be used in this special kind of steam turbine; practically any kind of fuel could be used as against the virtual necessity for fuel oil or gas in the case of the gas turbine, and for a highly refined and expensive light oil for the diesel engine.

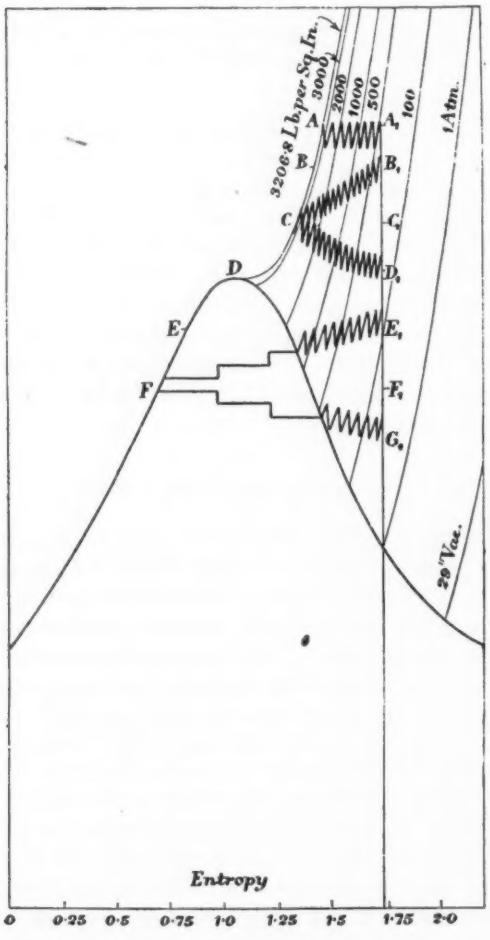

Fig. 3—Approach to reversible regeneration by adiabatic steps

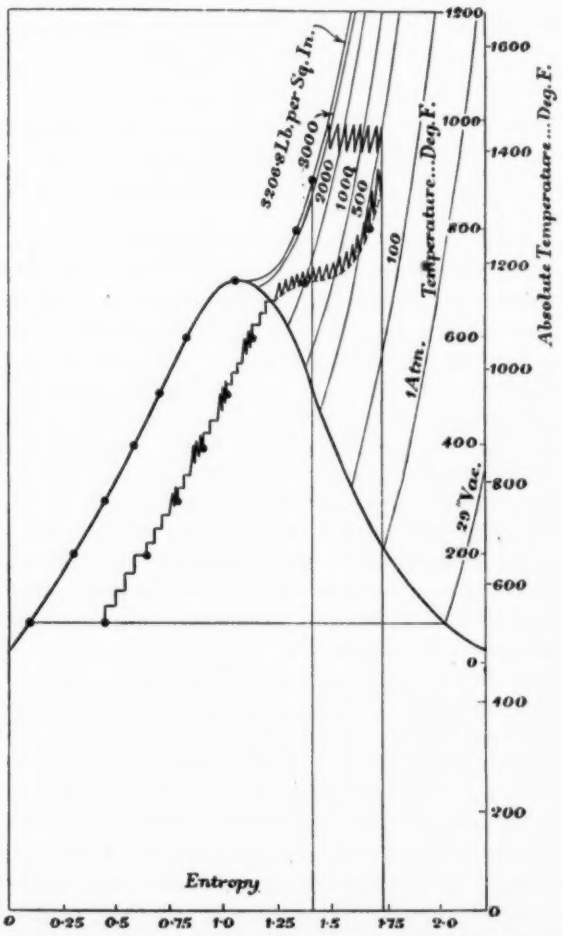

Fig. 4—Condition line of bled steam in Fig. 3 reduced to appropriate scale

COMPARISON OF PRIME MOVERS

Plant	Maximum Operating Temperature, Deg F	Ideal Heat Cycle Efficiency, Including Negative Work, Per Cent	Actual or Estimated Yearly Thermal Efficiency, "Units Sent Out" Basis, Per Cent	Ratio Actual/Ideal, Per Cent	Approx. Heat Rate of Turbine at most Efficient Point, Including Negative Work Btu per Kwhr	Ratio Actual/Ideal, Per Cent	Fuel
Diesel	4000–4500	....	30*	....	...	...	Diesel oil
Gas turbine	1000	....	15–16*	....	19,000*	...	Oil or gas
	1200	....	18–20	....	15,500	...	Oil or gas
Mercury-steam turbine (Kearny, U.S.A.)	About 1000	About 60	37.2†	62.0	...	...	Coal, oil or gas
Steam at 600 lb per sq in. (Battersea A)	850	44.3	27.63	62.4	9,650*	80.0*	Coal, oil or gas
Steam at 1250 lb per sq in. (Port Washington, U.S.A.)	825	48.7	32.2	66.2	8,850	79.3	Coal, oil or gas
Steam "Super-regenerator"	850	58.8	35–38*	60–65*	7,250*	80.0*	Coal, oil or gas
Steam at 1350 lb per sq in. (Battersea B)	950	49.3	30–31*	61–63*	8,700*	80.0*	Coal, oil or gas
Steam at 2300 lb per sq in. (Twin Branch, U.S.A.)	950	51.4	34	66.1	8,300*	80.0*	Coal, oil or gas
Steam "Super-regenerator"	950	61.7	37–40*	60–65*	6,900*	80.0*	Coal, oil or gas
Steam "Super-regenerator"	1000	63.0	38–41*	60–65*	6,800*	80.0*	Coal, oil or gas
Steam "Super-regenerator"	1200	67.7	40–43.5*	59–64*	6,400*	79.0*	Coal, oil or gas

* Estimated by author. † Short period run.

What advantages would the successful development of the "super-regenerator" have over the diesel engine for marine propulsion? For one thing, probably higher thermal efficiency, even at a limited upper temperature of 850 F, for which the machine could be constructed largely of mild steel. It should have the reliability and low maintenance costs of a steam-turbine plant, as compared with a reciprocating plant; and, finally, the ability to use powdered coal or boiler oil as against the expensive diesel oil. For land power production the thermal efficiency now envisaged should show a great saving on the present consumption of coal used in generating the nation's electricity.

The accompanying table sets forth various prime movers previously mentioned, for purposes of comparison, and the figures are what might be expected when producing electricity on a yearly variable-load basis. The thermal efficiency of the diesel engine for ship propulsion, fuel to mechanical power, would undoubtedly be better in favorable circumstances, and the special high-pressure steam-plant technique would show a considerable decline in thermal efficiency when adapted to ship propulsion, except in very large vessels where it might be economically justified. It is suggested that the 850 F steam "super-regenerator" plant would, however, prove of at least as good thermal efficiency as the diesel engine for ship propulsion, even in moderate sizes, with much lower costs of fuel and maintenance.

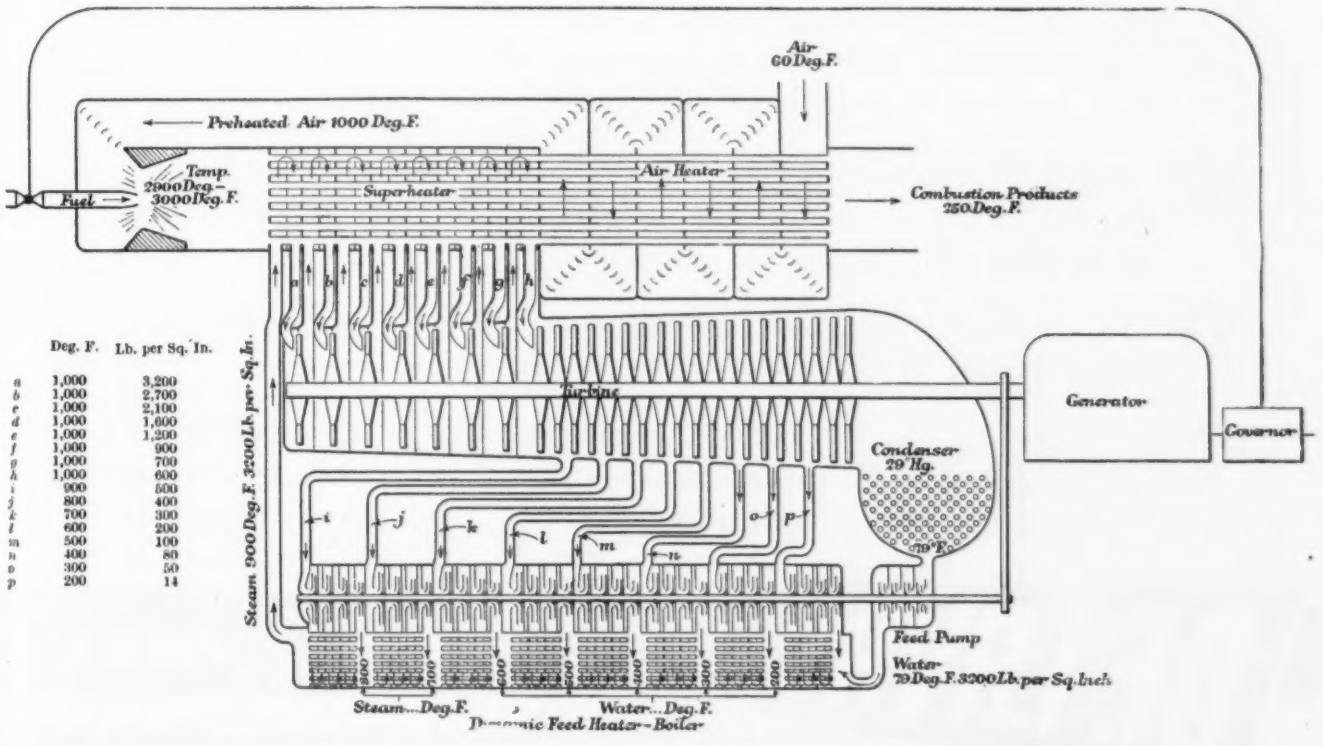

Fig. 5—Schematic diagram of "super-regenerator", employing turbine and multiple superheater with steam conditions as in Figs. 1–4

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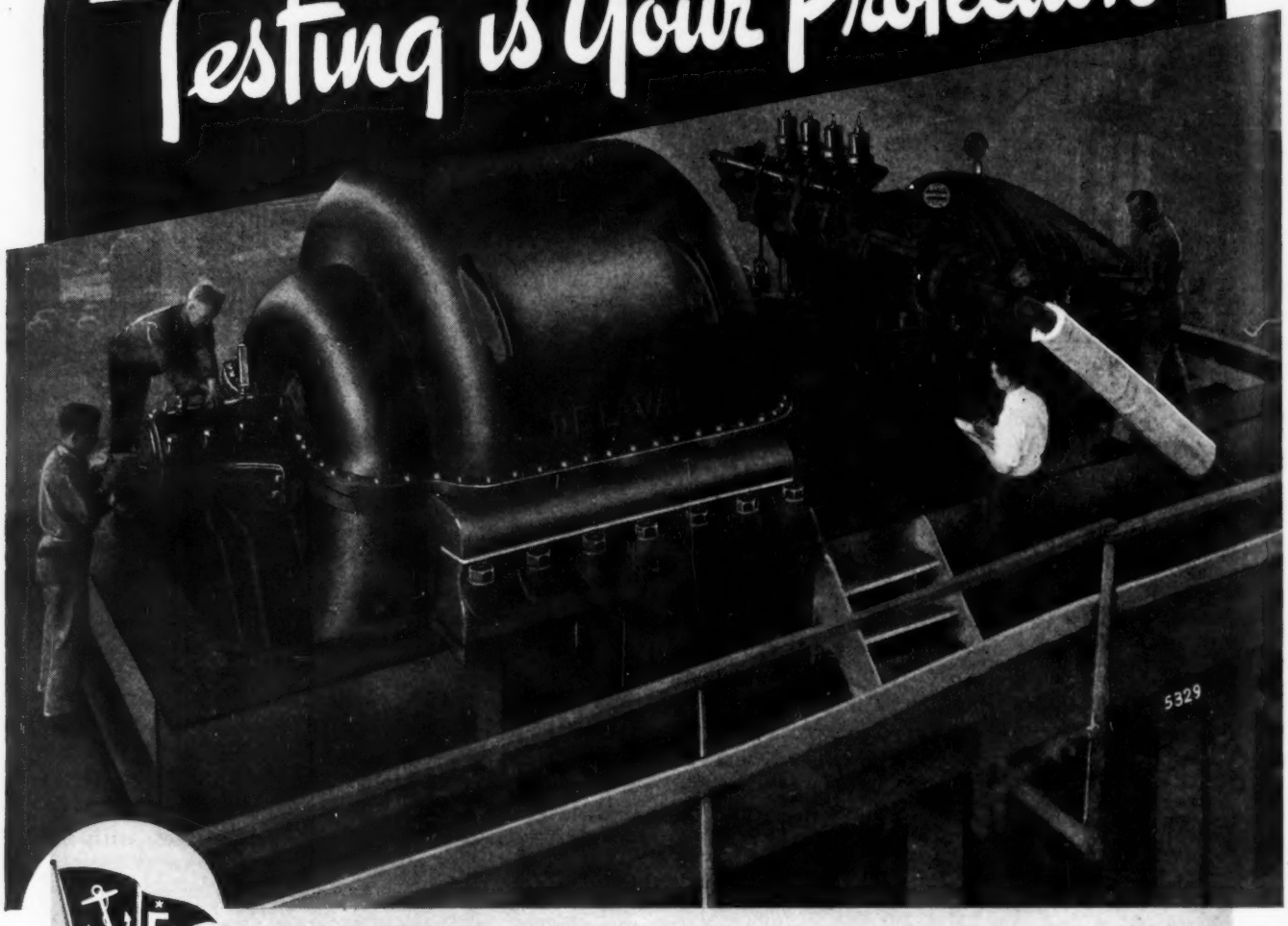

90,000 cu. ft. per min. turbine-driven blast furnace blower on test in De Laval Works

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Elevation of Dewpoint by Catalytic Action

Evidence is submitted to indicate that high dewpoint can be caused by increased temperature of the superheater tubes, the oxide film catalyzing the conversion of sulphur dioxide, formed by combustion, into sulphur trioxide, thereby increasing the sulphuric acid content of the flue gases. Coating the catalytic surfaces with lime or finely pulverized ash or dust at intervals promises to prevent the formation of acid and consequent corrosion of waste-heat recovery surfaces.

WHERE the dewpoint temperature of gases leaving a boiler is relatively high it may become impracticable to extract the optimum amount of heat by means of heat-recovery surface owing to corrosion of economizer and air heater surface brought about by the disposition of sulphuric acid from the presence of sulphur trioxide. The condition is more prevalent with stoker-fired units than those fired with pulverized coal and it has been aggravated in many cases by the necessity of burning low-grade coals having high sulphur content.

The problem was investigated some years ago by H. F. Johnstone at the University of Illinois Engineering Experiment Station, involving studies at several plants, both stoker and pulverized-coal fired, with analysis for SO_2 and SO_3 at various points in the path of the gases. The conclusions then reached were that very little catalytic oxidation of SO_2 to SO_3 occurs during the passage of the gases from the furnace to the stack and that the SO_3 is formed primarily in the furnace.

Inferior Coals Responsible for Increased Corrosion

Corrosion of air heaters and economizers has been accentuated in England during the war due to the necessity of burning coals of higher sulphur content, and additional light has been thrown on the problem by an investigation reported to The Institution of Mechanical Engineers (England) by W. F. Harlow[1] in a paper entitled "Causes of High Dewpoint Temperatures in Boiler Flue Gases."

Briefly, the author states that, as a result of accumulated experience indicating an increase in the dewpoint of flue gases during their passage through the boiler, the production of sulphur trioxide by catalytic action was suspected. It was therefore decided to test this by passing flue gases of normal dewpoint characteristics over the various metals present in the boiler passes when heated to temperatures likely to be attained in a modern plant. In the test apparatus the relative changes in dewpoint conditions were determined by the amount of acid formed on water-cooled elements having the same surface temperature. This method avoided difficult and controversial measurements of sulphur trioxide content and dewpoint temperatures and, at the same time gave a positive indication of the dew-forming properties of the gases.

It was found that rusty mild steel, superheater scales and firebrick, when raised to a sufficiently high temperature, were each capable of elevating the dewpoint of the flue gases, as indicated by an increase in acid formation. This was particularly evident in the case of rusty mild steel when heated to 1000 F. With this material, there was a small but gradually increasing amount of acid formed, even at comparatively low temperatures, which increased sharply at 800 F and up to 1100 F, after which there was a decrease. In fact, it has long been well known that iron oxide and refractory materials are catalytic to gaseous reactions.

Sulphur Dioxide Catalyzed to Sulphur Trioxide

Evidence is submitted to indicate that high dewpoint can be caused by increased temperature of the superheater tubes, the oxide film catalyzing the conversion of the sulphur dioxide, formed by combustion, into sulphur trioxide, thereby increasing the sulphuric acid content of the flue gases. Coating the catalytic surfaces, with lime or finely pulverized ash or dust at intervals promises to prevent the formation of acid and consequent corrosion of waste-heat recovery surfaces.

It follows that the elevation in dewpoint, due to an increase in sulphur trioxide, from sulphur dioxide, is brought about by passage of the gases over the heating surfaces of the boiler and particularly the superheater where temperatures from 800 to 1200 F exist. That is, the surface catalysis which takes place on the film of ferric oxide causes part of the sulphur dioxide to combine with additional oxygen to form sulphur trioxide. The increase in superheat temperature of modern practice is conducive to such action.

Study by the author of the steam conditions and operating experiences in a large number of plants indicated that such trouble is closely related to the degree of superheat, upon which the temperature of the exterior tube surface largely depends, although other variables are realized. One stoker-fired plant having a total steam temperature of 650 F was entirely free of such trouble whereas a later plant burning the same coal but operating at 850 F had been seriously affected. In still another plant with 950 F steam temperature the trouble had been particularly severe.

Attempts to avoid the formation of acid by preventing

[1] With International Combustion Ltd., Derby, Eng.

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oxidation of the steel by flue gases at high temperatures proved to be no simple matter. Nickel-plated steel, stainless steel, and steel sprayed with aluminum, all failed to resist oxidation and the resulting catalytic action.

Methods of coating the catalytic surfaces by inert material were carried out with more success. Spraying the catalyzing surfaces with milk of lime while they were in a heated condition was found to inhibit the acid formation. Also, spraying the surfaces with an aqueous suspension of pulverized-coal precipitator dust was found effective. With reference to the latter it was believed that the blinding effect of this extremely fine dust, much of which is below 10 microns, may be responsible for the comparative immunity of pulverized-coal-fired installations from this trouble. The length of time during which such spraying will provide immunity from acid formation has not been definitely determined, but it seems probable that periodic spraying would be necessary.

Black Magnetic Oxide Inhibits Corrosion

An additional important and interesting fact has emerged from these studies; namely, that when the flue gases are passed over steel which has been coated with black magnetic oxide (Fe_3O_4) such as is obtained by heating strongly in air, the catalytic action and acid formation is almost entirely inhibited over the whole temperature range. When, however, this black oxide has been removed by sand-blasting and the material is again subjected to the action of the flue gases, the characteristic red oxide (presumably Fe_2O_3) appears. This has the catalytic, acid-forming effects previously mentioned.

The inactivity of the black oxide may explain the immunity of new plants which is frequently observed, since new tubes are usually coated with black oxide as a result of their heat treatment during manufacture. The time interval necessary for this inactive layer to disappear and allow the red oxide to form may well represent the period during which such new plants are free from trouble of this sort.

Further tests were carried out to determine the effect of increasing the water-vapor content of the gases by steam injection. It was found that as the water vapor increased, the amount of acid greatly increased when the iron oxide catalyst was present, but when it was absent the injection of steam had only small effect.

The investigations are being conducted further, hence Mr. Harlow's paper may be regarded as a preliminary report.

Utility Taxes

Total taxes charged to electric operations of the larger (Classes A and B) privately-owned electric utilities in the United States for the second quarter of this year are reported by the Federal Power Commission to have been $183,206,000, as compared with $166,109,000 for the corresponding period last year. This represented an increase of 10.3 per cent. Of this total $124,463,000 represented federal taxes, an increase of 14.7 per cent over the second quarter of 1943, and $58,743,000 covered state and local taxes.

Smoke Prevention with Stoker Firing

Following is an abstract of a paper by J. W. Armour[1] before the 1944 Convention of the Smoke Abatement Association of America in which the author, from the standpoint of the stoker manufacturer, reviews some of the conditions responsible for smoke with underfeed stokers. He points out how they can be avoided or minimized, and thus lessen the task of the smoke abatement engineer.

THE stoker manufacturer desires installations that will operate smoothly and at high efficiency and if that is attained through proper engineering, the nstallation should comply with the rules of smokeless operation.

Sometimes, however, a stoker or other fuel-burning equipment is installed that doesn't have a chance to operate without causing a smoke violation. In some such cases the user is definitely responsible and in others responsibility rests with the stoker manufacturer. That is, in too many instances the purchaser of a stoker lets perhaps a dozen stoker manufacturers know that the cheapest price is going to get his order. With so many getting such an inquiry, there are likely to be one or two who will take a chance by skinning the job and making impossible guarantees in order to put in a low-priced stoker and get away with it.

Again, because of war conditions, there are many cases where plants have been ordered to change over from oil to coal. In many such instances the users have felt that such orders were unwarranted interference by the government and have determined to go through the motions with no intention of doing the job correctly and efficiently. They either do not, or refuse to, realize that the boiler capacity must be reduced with coal firing, unless material furnace changes are made. The result is trouble, smoke and condemnation of the stokers and stoker firing.

Some of the component parts of an installation that are pertinent from the standpoint of smoke abatement are stoker size, furnace dimensions, furnace construction and operation. The following discussion of these factors will concern mainly underfeed stokers, particularly of the industrial type.

Stoker Size

It is comparatively easy to select the right size of stoker if the amount and type of coal to be burned are known. An oversized stoker is not likely to contribute to smoke but one of undersize will do an excellent job of making smoke and contribute greatly to the fly-ash nuisance; so the purchaser must be sure of his requirements. One difficulty in putting in a stoker of sufficient size is sometimes the limitation imposed by the boiler. If there is not provision for proper grate area, a definite limit is imposed on the amount of coal that can be burned, hence on the maximum output of the boiler. This condition actually exists with some standard-size boilers and there are cases where the stoker manufacturer has been asked to provide equipment to produce ratings that are beyond the capacity of good stoker practice.

What is good stoker practice? This will depend upon the fuel, but considering a fuel such as Eastern Kentucky or West Virginia medium volatile coals, good practice represents a limit of about 35 lb per sq ft of grate surface for single-retort stokers and 45 lb for multiple-retort stokers. For smaller stokers 35 lb should be close to the top whereas with larger multiple-retort stokers, 45 lb is a good operating figure for maximum continuous load; although, under certain conditions, it can be exceeded. In exceptional cases ratings have been carried as high as 100 lb per sq ft on large central-station stokers, but the fly-ash carryover is large and collectors must be installed. High ratings mean high forced-draft pressures; and these, in turn, mean excessive fly ash.

Consideration should be given to the proportion of the stoker, especially of the single-retort type. A wide short stoker will not have the coal-burning capacity of a longer, narrower stoker of the same grate area because there will be less active surface in the former.

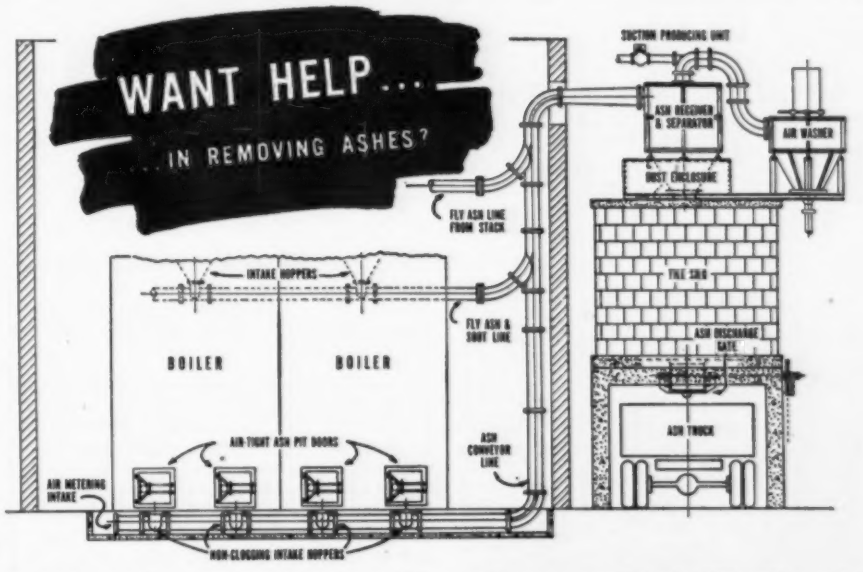
Furnace Design and Dimensions

It is possible to have ample volume without having a furnace of such shape that proper space or distance is provided for the completion of combustion. If combustion of the volatile gases is not completed before they reach a cold boiler surface, smoke will result, or if combustion of the coke particles is not completed before they reach a cold surface, fly ash will result.

On the other hand, it is possible to have too much distance to the boiler surface. A high brick furnace with boiler surface forming the roof may have ample volume and still not give good results. This is because the radiant heat absorption is inversely proportional to the square of the distance. A good rule to follow for furnace volume in stoker firing is a maximum heat release of 45,000 Btu per cu ft.

If sufficient boiler heating surface is not exposed to the furnace, the temperature

[1] Vice President, Riley Stoker Corporation.



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will build up to a point where the ash will slag and brickwork fail. On the other hand, if too much cold surface is exposed, the fuel-bed temperature is likely to be reduced to the point where proper combustion will not take place. With a furnace that is too hot, slag is formed and when the fireman bars the fire to get rid of clinker, smoke will go up the stack; whereas if it is too cold, the coal will not coke properly and when the fireman bars the fire to open up the fuel bed, smoke will be produced. A simple rule to follow for good results with stokers is a ratio of cold surface to that of the entire furnace of 0.19–0.22 to 1. The lower figure would apply to the higher ash-fusion coals and the higher figure to lower ash-fusion coals.

Today one is not sure of what kind of coal will be available, so it becomes necessary to try to design for the poorest grade that may be in the local market. In this connection, the smoke abatement engineer should have a good idea of the fuels that are available now and are likely to be available in the future and should insist on correct conditions before approving an installation.

Operation

A stoker is a machine containing some finely machined parts; hence dirt should not be allowed to accumulate around the drive and the motor.

Next comes proper feeding of the coal. Where for some reason coal has hung up in the hopper, or it has been allowed to run empty, re-establishment of the supply means feeding an excess to build up the fuel bed to normal. But if the coal is fed more rapidly than air is supplied smoke will result. Therefore, it is most important to keep the hopper filled and make sure that the coal is being fed properly to the stoker.

After the coal has been supplied to the stoker, it must be properly distributed. Many stokers are supplied with auxiliary rams or pushers which are adjustable and where the type of coal changes adjustments should be made. Poor distribution means working the fire with the slice bar and rake which, in turn, means rapid release of volatiles and smoke.

The best type of automatic control is that which follows steam demands and maintains a constant steam pressure regardless of load; but it is most important that it be maintained in proper operating condition, for hunting will contribute to smoke. Where the start-stop type of control is employed it should be applied with judgment, and with this type the maximum fuel-burning rate should be low, not over 20 to 25 lb per sq ft, and the air and fuel feed should be properly proportioned. If the stoker is allowed to run at full blast during the "on" period, the fuel-burning rate will be too high, large quantities of gas will be liberated and the furnace overloaded with resulting fly ash and smoke. Too often the operator thinks he is saving money if the stoker runs for a short time and is shut off for a long time, but he is actually wasting money through loss in efficiency. In other words, the start-stop control will give best results if the peaks and valleys can be smoothed out and the stoker operated at moderate ratings.

Utility Peaks and Output

Peak demands of the principal electric utility systems of the country for July 1944 aggregated 35,898,274 kw, an increase of 55,000 kw over the preceding month and a gain of 4.8 per cent over July 1943. These figures are reported by the Federal Power Commission which also gives the corresponding production of electric energy for public use during July 1944 as 18,726,192,000 kwhr, which represents an increase of 1.7 per cent over that for the same month of 1943. Of the total, 31.9 per cent was produced by water power.

The same source reports 6,713,188 tons of coal consumed in July 1944 and that coal stocks were 8.9 per cent less than that of the same date last year. Of the total coal consumed, 95.6 per cent was bituminous. Fuel oil consumed by the utilities decreased 6.8 per cent from that of June and natural gas increased 7.5 per cent.

Nation-Wide Association of Engineers Favored

A nation-wide association of engineers, embracing all branches of that profession, was advocated recently by C. A. Powel, newly elected president of the American Institute of Electrical Engineers. Speaking before the Los Angeles Section Meeting of the A.I.E.E., Mr. Powel declared that "in a world where scientific knowledge and engineering achievement are shaping our habits of thought and life, there is urgent need for an all-inclusive organization that can speak for engineers in the councils of government and the ranks of industry. The desire for collective action is as powerful among engineers as it is among members of labor or white-collar unions. What form such collective action should take is the main question facing members of the profession today."

Two plans of joint action have already been tried, he reported. The first, put into effect by the American Society of Civil Engineers, authorizes local sections of the A.S.C.E. to set themselves up as collective bargaining units. The second plan favors the inclusion of engineers in the general white-collar unions.

"Both of these plans offer serious difficulties," Mr. Powel asserted. "The first would probably mean the elimination of supervisory workers from the bargaining units and, in addition, present a number of legal obstacles. The second plan has several disadvantages. In any concern regardless of size, the ratio of engineers to draftsmen, clerks, accountants and others is of necessity small. In addition, the engineer would be subjected to decisions and rules which are clearly intended for a great majority of men doing an entirely different class of work."

He disclosed that the A.I.E.E. had formed a joint committee with the Ameri-

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can Society of Mechanical Engineers to investigate the subject of collective bargaining units for local Society sections, and added that "their preliminary study has brought out the tremendous difficulties of such a plan."

Mr. Powel declared that by the very nature of his work the engineer has something different to sell than the machinist, the clerk or the laborer. His training has developed his creative instinct and his work is not adapted to control by any ordinary time-study procedure, nor can his hours of work be put on a rigid schedule. He is much more likely to become an individualist than a man working with a group in the shop. Nevertheless, he has the same desire to seek protection in collective action.

"It is doubtful," Mr. Powel concluded, "if a bargaining agency in the ordinary sense of the term is necessary or desirable for engineers. The protection he rightly seeks could probably be provided more effectively by an overriding association embracing all engineers devoted specifically to the welfare of the profession as a whole and the individual members comprising it."

Welding Society to Hold 25th Annual Meeting

Seventeen technical sessions embracing more than sixty papers on welding subjects will feature the 25th Annual Meeting of the American Welding Society to be held in the Hotel Cleveland, Cleveland, Ohio, from October 16 to 19. The meeting is being held, as in past years, in conjunction with the National Metal Congress.

Three talks emphasizing the importance of welding in meeting the wartime production needs of the nation will be delivered at the opening session on the morning of October 16. Admiral H. L. Vickery, U. S. Maritime Commission, will speak concerning the use of welding in shipbuilding; Col. S. B. Ritchie, U. S. Army, of its use in the fabrication of ordnance equipment; and, W. B. Stout, Consolidated Vultee Aircraft Corporation, of welding in aircraft production. The work of the American Welding Society in promoting the production of needed ordnance equipment will be recognized by the presentation of the Ordnance Distinguished Service Award at this session.

The technical papers to be presented, of which there will be over fifty, will cover the entire range of application of welding and applied processes, emphasizing wartime applications, as well as research. Most of these applications and developments will be of importance when industry returns to normal peacetime production.

A session Wednesday afternoon on Piping and Pressure Vessels will comprise the following papers: "Pressure Vessel Welding," by E. B. McGuire; "Normalizing of Welds in Carbon-Molybdenum Steel Pipe by 60-Cycle Induction Heating," by I. A. Rohrig and D. H. Corey; "Properties of Welded Joints Between Dissimilar Metals," by E. C. Chapman and R. E. Lorentz; and "Oxy-Acetylene Pressure Welding," by A. R. Lytle.

Books

1—ASTM Specifications for Steel Piping Materials

255 pages 6 × 9 Price: $1.75

The 1943 edition of this compilation of all specifications issued by the A.S.T.M. covering steel piping materials includes 44 standards and provides 25 emergency alternate provisions that have been issued to aid in expediting procurement. Many of the specifications and emergency provisions are part of schedules appended to WPB limitation order L211.

2—X-Rays in Research and Industry

By H. Hirst

130 pages $4^3/_4 \times 7^1/_2$ Price: $2.50

The book offers a thorough and self-contained study of the principles of X-ray technique and the versatility of its application. Chapter headings are as follows: Production of X-Rays; Properties of X-Radiation; Structure of Crystals; Methods of Crystallographic Examination; Applications of X-Ray Methods; and Industrial Radiography. The text is cogent and to the point, and the use of small type allows a great deal to be said in the 130 pages which comprise the book which is admirably illustrated.

3—Mathematics Dictionary (Revised Edition)

By Glen James and Robert C. James

320 pages $6^1/_2 \times 9^1/_4$ Price $3.00

This invaluable reference book, which first appeared in 1942, is designed to provide a condensed source of facts and principles for men working in practical fields where mathematics is an essential feature. In this revised edition, many new terms have been added, more "working examples" have been introduced, irregular plurals added, typographical errors corrected, five-place logarithm tables have been substituted for four-place tables and extensive integral tables have also been added. Elementary concepts and definitions have been simplified for the benefit of the layman, but basic terms beyond calculus have been included so that the dictionary contains most of the terms which form the foundation of any advanced course in mathematics.

4—How to Maintain Electric Equipment in Industry

372 pages $8 \times 10^1/_2$ Price: $1.75

The procedures outlined in this substantial manual, published by General Electric Company, are intended to help industrial plants get maximum performance and longest life out of their General Electric equipment through proper upkeep.

The volume comprises 20 sections on How to Maintain: Battery Trucks; Bushings; Cable and Wire; Capacitors; Diesel-Electric Locomotives; Electric Furnaces; Fuse Cutouts (Distribution Type); Industrial Control; Instruments; Lighting; Lightning Arresters; Midget Heaters; Mine Locomotives; Motors and Generators; Rectifiers; Regulators; Switchgear; Transformers; Turbines; and Welders; also sections on Insulation; Technical Data; Renewal Parts; and G-E Service Shops. Preventive-maintenance schedules and trouble-shooting charts are included in nearly all types of apparatus listed.

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